

SOLAR POWER PLANTS AND THEIR APPLICATION

Features of Working Process in Stirling Engines with Direct and Reverse Cycles

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Abstract—The features of processes in thermodynamic transformers operating with direct and reverse Stirling cycles are considered. It is shown that nonstationarity of the heat exchange and gas dynamic processes influences the cycle efficiency. The influence of nonuniformity of solar radiant flux input to the heat absorber of the Stirling engine on solar energy transformation efficiency is evaluated.

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The features of thermodynamic processes in apparatus operating with direct and reverse Stirling cycles were considered in [1–3]. It is known that during the operating cycle with frequency of 25 to 50 Hz (1500 to 3000 rpm) the velocity and direction of the working body motion, its temperature, pressure, and other thermophysical characteristics change, i.e., the flow of the working body in the operating loop is nonstationary and has a pulsing nature. In designing the Stirling engines, the thermodynamic calculations of the working loop were performed in the quasi-stationary approximation using the criterial equations for stationary flow, which led to significant errors. With account for this, the program for calculating the Stirling engine characteristics, based on the Schmidt isothermal model, was improved [1–3], and the piston sinusoidal motion was replaced by the equations of piston real motion with account for the specific drive mechanism.

In the present work the results of computer simulation of a solar Stirling engine with 5 kW power and "swashplate" type drive are presented. The formulas for calculating the change of the working body masses in the working and heat-exchange spaces, and also the features of the computer program algorithm are given in [1].

Figure 1 shows the calculated dependences of the distribution of the working body (helium) masses on the engine shaft rotation angle with average cycle pressure 100 bar.

Change of the helium masses in the engine working spaces (hot and cold) is caused by motion of two adjacent pistons, moving with phase shift 90° and forming a single working loop, i.e. a single engine. Change of the volumes of the hot and cold spaces, helium pumping from the hot space to the cold space through the heater, regenerator, and cooler and vice versa takes place, which causes change of the working body temperature and pressure in the engine. Since the engine is double-sided, the four pistons form four identical work-

ing paths. Change of the helium masses in spaces of the heat exchangers (heater, regenerator, and cooler) depends on change of helium pressure during the cycle. When the piston adjacent to the cold space is at bottom dead center, the pressure of the working body passes through a maximal value and the expansion process occurs. The amount of the working body in the heat exchanger spaces is maximal with maximal helium pressure in the cycle. The minimal value of the pressure in the cycle is achieved at the end of the expansion process, i.e. at bottom dead center of the piston adjacent to the hot space of this loop. The amount of the working body in the heat exchanger spaces is minimal with minimal helium pressure in the cycle.

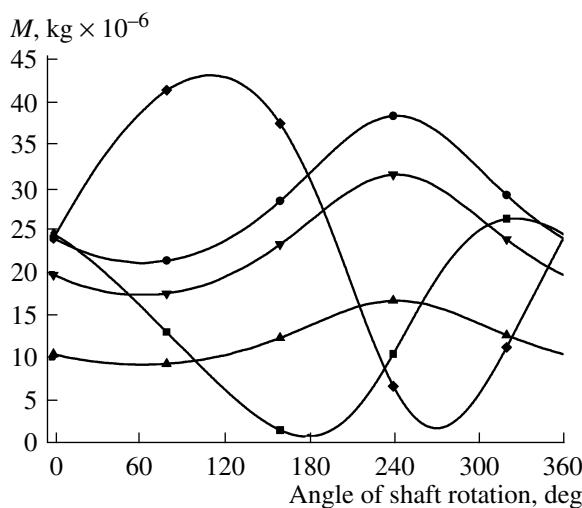


Fig. 1. Change of working body masses in working spaces of Stirling engine: ■—hot space; ●—heater; ▲—regenerator; ▼—cooler; ◆—cold space.

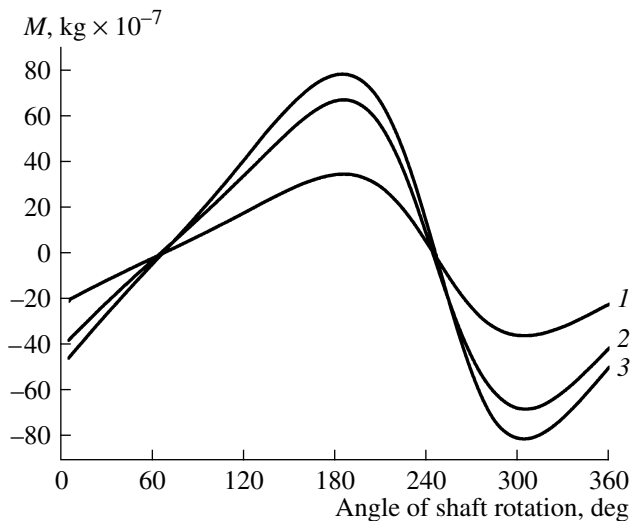


Fig. 2. Change of working body masses in heater, regenerator and cooler as function of engine shaft rotation angle: 1—regenerator; 2—cooler; 3—heater.

Change of the working body masses in the heat exchangers is determined by the nature of the motion and intensity of heat exchange between the working body, i.e. helium, and the heat exchanger wall or regenerator headpiece. The nature of this motion is nonstationary, the direction of motion changes twice per cycle. According to the experimental data, for the laminar region the equations of stationary heat exchange yield understated coefficients of heat exchange in the case of the pulsing flow regime [4].

In the Stirling engine heat exchangers the flow regime changes constantly from laminar to turbulent, passing through the transient flow region. Experimental investigations of these regimes have not been carried out. To improve the equations of stationary heat exchange, experimental investigations of heat exchange under conditions of the working regimes of the Stirling engine and the Stirling refrigerating machine are required. The velocity regimes of the working body flow in Stirling engine heat exchangers can be explored by computer simulation [1].

As illustrated in Fig. 2, the maximal change of the magnitude of the masses occurs approximately with maximal velocity of the piston adjacent to the cold space. This is because the gas temperature in the cold space is minimal, and therefore the density is maximal. Similar dependences (see Fig. 3) were obtained for the mass flow rates, considering the cycle frequency (25 Hz or 150 rpm).

The calculation of the heat transfer coefficients in the heat exchangers was performed in the quasi-stationary approximation [1]. The formulas for calculating the instantaneous and cycle-average coefficients of heat transfer in the refrigerator and heater are presented in [1, 5].

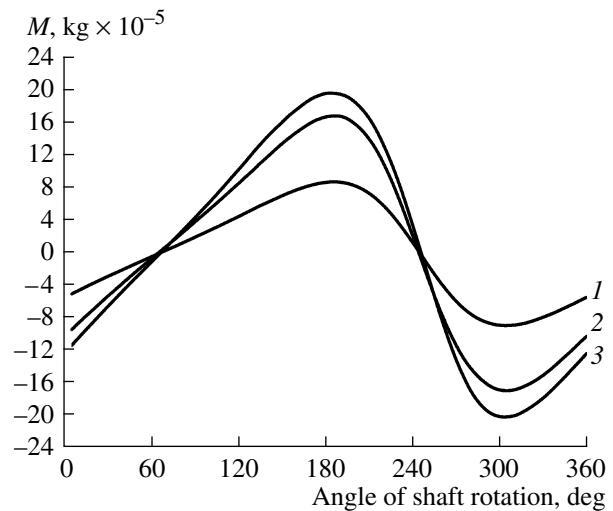


Fig. 3. Dependences of working body mass flow rates in heater, regenerator, and cooler: 1—cooler; 2—regenerator; 3—heater.

The results of the calculations are presented in Fig. 4. The nonuniformity of the radiant flux density in the focal spot, existing even in the ideal paraboloid concentrator case, also depends on the quality of the reflecting surface and its shape. This leads to nonuniformity of the density of the heat flux incident on the heater external surface and nonuniformity of the formed temperature field of the heater surface. In its turn, the induced convective heat exchange between the internal heater surface and the pulsing flow of the working body causes oscillations of the heater wall temperature with the frequency of the pulsing flow.

The degree of heater temperature field nonuniformity is an important factor influencing the efficiency of solar energy transformation. The cycle efficiency of any heat engine depends on the temperatures of heat supply and removal. At the same time the maximal admissible heater temperature depends on its material. For 1X18H10T stainless steel, used for heater fabrication, the limiting long-term strength temperature is about 650°C. The temperature field nonuniformity reduces the average temperature of the heater surface in comparison with the maximum possible temperature based on the material properties and decreases the cycle efficiency. Therefore, one of the design problems is to study the heat supply processes and, correspondingly, the dependences of the degree of nonuniformity of the temperature field of the surface of the heat absorber and heater on the nonuniformity of the distribution of the heat flux to the heat absorber during heat removal by the nonstationary pulsing flow of the working body.

For the ideal case the efficiency of the Stirling cycle, and for the Carnot cycle, is determined by the relation

$$\eta = 1 - \frac{T_X}{T_\Gamma} \quad (1)$$

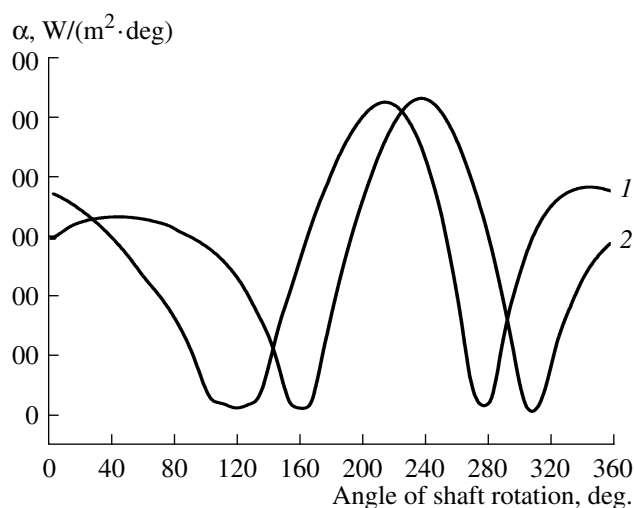


Fig. 4. Dependence of heat transfer coefficient on shaft rotation angle: 1—heater; 2—cooler.

In the nonuniform heat supply case, the temperature of the surface of the heat absorber or heater of the Stirling engine will also be nonuniform and depend on the heat removal nonuniformity. In this case the equation for the efficiency will be:

$$\eta = 1 - \frac{T_x}{T_\Gamma - \frac{\Delta T_\Gamma}{2}}, \quad (2)$$

where ΔT_Γ is the maximal value of the difference of the temperatures on the heat absorber surface. The dependence of the ideal Stirling cycle efficiency on the value of the nonuniformity of the heat absorber temperatures is shown in Fig. 5.

Figure 5 shows that the nonuniformity of the temperatures on the heat absorber surface causes decrease of the cycle efficiency. On the other hand, cyclic change of the coefficients of heat transfer from the internal wall of the heat absorber to the working body causes cyclic variation of the temperatures of the internal and external walls of the heat absorber.

To refine the influence of these factors, experimental and theoretical investigations of the heat exchange pro-

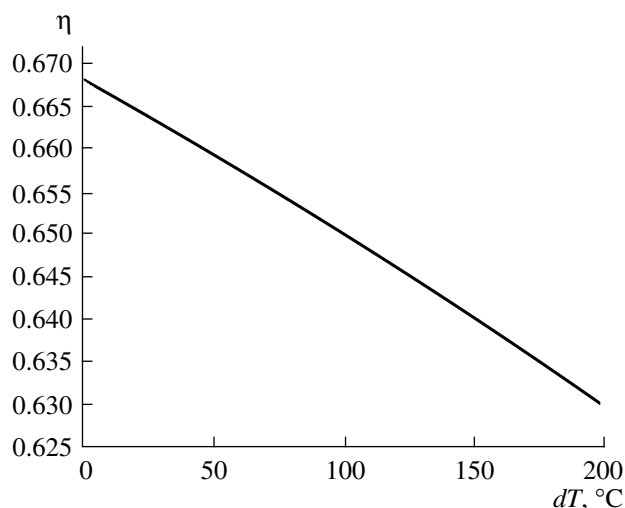


Fig. 5. Dependence of ideal Stirling cycle efficiency on magnitude of nonuniformity of heat absorber temperature.

cesses in the heater and cooler of the Stirling engine are required.

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